SPECIFICATION

OBLIQUE CONTACT DOUBLE ROW BALL BEARING AND
METHOD OF IMPARTING PRELOAD IN THE BALL BEARING

FIELD OF THE INVENTION

[1]　　The present invention relates to an oblique contact double row ball bearing for supporting a pinion shaft of a differential device additionally provided in a vehicle or the like under a free rotation, more specifically to an oblique contact double row ball bearing in which pitch circle diameters of double rows are different to each other, in other words, raceway diameters of the double rows are different to each other, and a method of imparting a preload to the ball bearing.

BACKGROUND OF THE INVENTION

[2]　　A tapered roller bearing is used as a roller bearing for supporting a pinion shaft of a differential device additionally provided in a vehicle, or the like under a free rotation. The tapered roller bearing has a large load capacity, however, its rotation torque is large. Therefore, an oblique contact ball bearing (angular contact ball bearing) may be incorporated into the differential device or the like in place of the tapered roller bearing (for example, see the Patent Document 1), or an oblique contact double row ball bearing called a tandem double row ball bearing in which pitch circle diameters of double rows are different to each other, in other words, raceway diameters of the double rows are different to each other, may occasionally be incorporated into the differential device or the like.

[3]　　The oblique contact double row ball bearing having the pitch circle diameters different to each other is effectively used particularly for the opinion shaft of the differential

device or the like because the rotation torque thereof is smaller than that of the tapered roller bearing and the load capacity is sufficiently large.

[4] In the case of incorporating these bearings into the differential device, the bearings are managed and stored in a state where a preload as prescribed is applied thereto.

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DISCLOSURE OF THE INVENTION

PROBLEM TO BE SOLVED BY THE INVENTION

[5] The preload is controlled (adjusted) in the bearing through the measurement of the rotation torque of the bearing. Therefore, a range where the preload is set is increased as the rotation torque is larger, which facilitates the adjustment. As described earlier, the tapered roller bearing is advantageous in its large load capacity and at the same time the rotation torque thereof is large. As a result, the adjustment range of the preload to be set with respect to the bearing is large, which makes it easy to control the preload. However, in the oblique contact ball bearing, which has the structure of the ball bearing, the rotation torque is small and the adjustment range of the preload to be set with respect to the bearing is thereby reduced. As a result, it is difficult for the preload to be set with a high accuracy.

[6] A main object of the present invention is to facilitate the control of the preload in the oblique contact ball bearing.

MEANS FOR SOLVING THE PROBLEM

[7] In order to achieve the foregoing object, an oblique contact double row ball bearing according to the present invention comprises double rows of balls axially interposed in raceways of inner and outer rings, wherein an inner clearance

between the balls in one of the rows and the raceway in which the balls are rolled, and an inner clearance between the balls in the other row and the raceway in which the balls are rolled are different to each other.

5   [8]   In a method of imparting a preload to the oblique contact double row ball bearing according to the present invention, the inner clearance between the balls in one of the rows and the raceway of the inner and outer rings corresponding to the one of the rows in which the balls are rolled, and the inner

10 clearance between the balls in the other row and the raceway of the inner and outer rings corresponding to the other row in which the balls are rolled, are set to be different to each other, and then a load is given to the inner and outer rings so that the inner clearances are sequentially reduced in order

15 to provide the preload to the inner and outer rings.

[9]   In the oblique contact double row ball bearing according to the present invention, any one of the clearances may be reduced earlier than the other.

[10]   The preload to be imparted to the bearing is generally

20 obtained through measurement of a rotation torque. The case of imparting the preload to the oblique contact double row ball bearing is considered here. In this case, when a thrust load S to be imparted to the inner and outer rings is hypothetically a "S2" value, comparison of an adjustment range "T1" of a rotation

25 torque T in a conventional oblique contact double row ball bearing to an adjustment range "T2" of the rotation torque T in the oblique contact double row ball bearing according to the present invention, which correspond to the "S2" value, it becomes T2 > T1. Therefore, when it is tried to obtain the

30 same preload, the preload can be adjusted in the range wider in the oblique contact ball bearing according to the present invention than that of the conventional oblique contact ball bearing, which consequently makes it easy to impart the preload

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with a high accuracy.

[11]   In addition, the preload may be set while the thrust load "S2" is being adjusted in the range of [S1] - [S3] in view of its tolerance in setting the preload. Such a case is considered. Comparing an adjustment range [T3] of the rotation torque T in the conventional bearing to an adjustment range [T4] of the rotation torque T in the bearing according to the present invention, T4 > T3 is obtained. When it is tried to obtain the same preload, the adjustment range of the rotation torque T (in other words, adjustment range of the preload) is increased in the bearing according to the present invention in comparison to that of the conventional bearing. As a result, the preload can be easily and accurately imparted.

EFFECT OF THE INVENTION

[12]   According to the present invention, the preload can be adjusted in the adjustment range wider than that of the conventional oblique contact bearing by making the rotation torque large, and the preload can be thereby accurately and easily imparted.

BRIEF DESCRIPTION OF THE DRAWINGS

[13]   Fig. 1 is a sectional view illustrating a schematic constitution of a differential device according to a preferred embodiment of the present invention.

Fig. 2 is a sectional view in which double row ball bearings in the differential device are enlarged.

Fig. 3 is a sectional view in which the double row ball bearings are further enlarged.

Fig. 4 is a sectional view illustrating a state where the double row ball bearings are being assembled.

Fig. 5 is a graph showing a relationship between a thrust load and a rotation torque.

DESCRIPTION OF REFERENCE SYMBOLS

[14]   1   differential device

       2   differential case

       6   pinion gear

       7   pinion shaft

       10   first double row ball bearing

       25   second double row ball bearing

       11   first outer ring

       21   first assembly component

       13   first inner ring

       12   second outer ring

       22   second assembly component

       14   second inner ring

       28,29   row of balls

       30,31   balls

PREFERRED EMBODIMENT OF THE PRESENT INVENTION

[15]   A preferred embodiment of the present invention is described referring to the drawings. Fig. 1 is a sectional view illustrating a schematic constitution of a differential device. Fig. 2 is a sectional view in which double row ball bearings are enlarged.

[16]   As shown in Fig. 1, a differential device 1 comprises a differential case 2. The differential case 2 comprises a front case 3 and a rear case 4. These cases 3 and 4 are coupled to each other by a bolt / nut 2a so as to thereby be integrated. Annular walls 27A and 27B in which ball bearings are applied are formed in the front case 3.

[17]   The differential case 2 comprises internally a differential speed-change mechanism 5 for differentially gearing right and left wheels, and a pinion shaft (drive pinion) 7 having a pinion gear 6 on one side thereof. The pinion gear

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6 is meshed with a ring gear 8 of the differential speed-change mechanism 5. A shaft part 9 of the pinion shaft 7 is formed in a stepwise shape so that a diameter thereof is gradually reduced on the other side than one side thereof.

[18]   The one side of the shaft part 9 of the pinion shaft 7 is supported by the annular wall 27A of the front case 3 so as to freely rotate around an axial center via a first double row ball bearing 10. The other side of the shaft part 9 of the pinion shaft 7 is supported by the annular wall 27B of the front case 3 so as to freely rotate around the axial center via a second double row ball bearing 25.

[19]   As shown in Fig. 2, the first double row ball bearing 10 is an oblique contact double row ball bearing, and comprises a single first outer ring 11 fitted to an inner peripheral surface of the annular wall 27A and a first assembly component 21. The first assembly component 21 is assembled into the first outer ring 11 from the pinion-gear side toward the opposite side of the pinion gear 6 (hereinafter, referred to as counter-pinion-gear side) along an axial direction so as to thereby constitute the first double row ball bearing 10.

[20]   The first outer ring 11 has a structure of a counterbored outer ring. More specifically, the first outer ring 11 comprises a large diameter outer ring raceway 11a on the pinion-gear side and a small diameter outer ring raceway 11b on the counter-pinion-gear side. A planar part 11c is formed between the large diameter outer ring raceway 11a and the small diameter outer ring raceway 11b. The planar part 11c has a diameter larger than that of the small diameter outer ring raceway 11b and continuous to the large diameter outer ring raceway 11a. An inner peripheral surface of the first outer ring 11 is thus formed in the stepwise shape.

[21]   The first assembly component 21 comprises a single first inner ring 13, a large-diameter-side row of balls 15, a

small-diameter-side row of balls 16, and retainers 19 and 20. The first inner ring 13 has a structure of a counterbored inner ring. More specifically, the first inner ring 13 comprises a large diameter inner ring raceway 13a and a small diameter

5   inner ring raceway 13b. The large diameter inner ring raceway 13a faces the large diameter outer ring raceway 11a in a radial direction. The small diameter inner ring raceway 13b faces the small diameter outer ring raceway 11b in a radial direction. A planar part 13c is formed between the large diameter inner

10   ring raceway 13a and the small diameter inner ring raceway 13b. The planar part 13c has a diameter larger than that of the small diameter inner ring raceway 13b and continuous to the large diameter inner ring raceway 13a. An outer peripheral surface of the first inner ring 13 is thus formed in the stepwise shape.

15   [22]   The large-diameter-side row of balls 15 are fitted to place on the pinion-gear side, in other words, between the large diameter outer ring raceway 11a and the large diameter inner ring raceway 13a. The small-diameter-side row of balls 16 are fitted to place on the counter-pinion-gear side, in other words,

20   between the small diameter outer ring raceway 11b and the small diameter inner ring raceway 13b.

[23]   In the first double row ball bearing 10, a contact angle of the row of balls 15 and a contact angle of the row of balls 16 have a same direction. In other words, a line of action

25   γ1 in accordance with the contact angle of the row of balls 15 and a line of action γ2 in accordance with the contact angle of the row of balls 16 face each other in a such a direction that an angle θ1 (not shown) made by the lines of action γ 1 and γ2 is 0° or an acute angle (0° $\leqq \theta$1<90°). Such a

30   constitution is adopted so that a preload is imparted to the both rows of balls 15 and 16 in a same direction (in the present case, direction from the pinion-gear side toward the counter-pinion-gear side). Further, the lines of action γ

1 and $\gamma2$ are tilted in such a direction that outer-diameter sides thereof are on the counter-pinion-gear side and inner-diameter sides thereof are on the pinion-gear side with respect to a thrust surface. To be brief, the lines of action $\gamma1$ and $\gamma2$ are tilted in the upper-right direction in Fig. 2 and FIG. 3. The retainers 19 and 20 retain balls 17 and 18 respectively constituting the rows of balls 15 and 16 at a position with circumferentially equal interval.

[24]   The pinion shaft 17 is inserted through the first inner ring 13, and an end surface of the first inner ring 13 abuts an end surface of the pinion gear 6 from the axial-center direction. The first inner ring 13 is sandwiched from the axial-center direction between the end surface of the pinion gear 6 and a plastic spacer 23 externally mounted on the shaft part 9 of the pinion shaft 7 at an intermediate position thereof for setting the preload.

[25]   In the first double row ball bearing 10, a diameter of the ball 17 in the large-diameter-side row of balls 15 and a diameter of the ball 18 in the small-diameter-side row of balls 16 are equal to each other, while pitch circle diameters D1 and D2 of the respective rows of balls 15 and 16 are different to each other. More specifically, the pitch circle diameter D1 of the large-diameter-side row of balls 15 is set to a value larger than that of the pitch circle diameter D2 of the small-diameter-side row of balls 16. As described, the first double row ball bearing 10 has a double row structure (rows of balls 15 and 16) in which the two rows of balls have the different pitch circle diameters D1 and D2 each other.

[26]   As shown in an enlarged view of Fig. 3, the balls 17 of the large-diameter-side row of balls 15 are placed so as to space at a predetermined radial clearance $\alpha1$ between the large diameter outer ring raceway 11a and the large diameter inner ring raceway 13a in an initial state before assembling

into the differential device. The balls 18 of the small-diameter-side row of balls 16 are placed so as to space at a predetermined radial clearance $\beta 1$ smaller than the radial clearance $\alpha 1$ ($\alpha 1 > \beta 1$) between the small diameter outer ring raceway 11b and the small diameter inner ring raceway 13b in the initial state before assembling into the differential device.

[27] The second double row ball bearing 25 is an oblique contact double row ball bearing, and comprises a single second outer ring 12 fitted to an inner peripheral surface of the annular wall 27B and a second assembly component 22. The second assembly component 22 is assembled into the second outer ring 12 from the counter-pinion-gear side toward the pinion-gear side along the axial-center direction.

[28] The second outer ring 12 has a structure of a counterbored outer ring. More specifically, the second outer ring 12 comprises a small diameter outer ring raceway 12a on the pinion-gear side and a large diameter outer ring raceway 12b on the counter-pinion-gear side. A planar part 12c is formed between the small diameter outer ring raceway 12a and the large diameter outer ring raceway 12b. The planar part 12c has a diameter larger than that of the small diameter outer ring raceway 12b and continuous to the large diameter outer ring raceway 12a. Accordingly, an inner peripheral surface of the second outer ring 12 is thus formed in the stepwise shape.

[29] The second assembly component 22 comprises a single second inner ring 14, a small-diameter-side row of balls 28, a large-diameter-side row of balls 29, and retainers 32 and 33. The second inner ring 14 has a structure of a counterbored inner ring. More specifically, the second inner ring 14 comprises a small diameter inner ring raceway 14a and a large diameter inner ring raceway 14b. The small diameter inner ring raceway 14a faces the small diameter outer ring raceway 12a

in a radial direction. The large diameter inner ring raceway 14b faces the large diameter outer ring raceway 12b in a radial direction. A planar part 14c is formed between the small diameter inner ring raceway 14a and the large diameter inner ring raceway 14b. The planar part 14c has a diameter smaller than that of the large diameter inner ring raceway 14b and continuous to the small diameter inner ring raceway 14a. An outer peripheral surface of the first inner ring 14 is thus formed in the stepwise shape.

[30]   The pinion shaft 7 is inserted through the second inner ring 14. The second inner ring 14 is sandwiched from the axial-center direction between the plastic spacer 23 for setting the preload and a shield 37.

[31]   The small-diameter-side row of balls 28 are placed to fit on the pinion-gear side, that is, between the small diameter outer ring raceway 12a and the small diameter inner ring raceway 14a. The large-diameter-side row of balls 29 are placed to fit on the counter-pinion-gear side, that is, between the large diameter outer ring raceway 12b and the large diameter inner ring raceway 14b.

[32]   In the second double row ball bearing 25, a contact angle of the row of balls 28 and a contact angle of the row of balls 29 have a same direction. In other words, a line of action γ3 in accordance with the contact angle of the row of balls 28 and a line of action γ4 in accordance with the contact angle of the row of balls 29 face each other in a such a direction that an angle θ2 (not shown) made by the lines of action γ3 and γ4 is 0° or an acute angle (0° $\leqq \theta$2 < 90°). Such a constitution is adopted so that the preload is imparted to the both rows of balls 28 and 29 in a same direction (in the present case, direction from the counter-pinion-gear side toward the pinion-gear side). Further, the lines of action γ3 and γ4 are tilted in such a direction that outer-diameter sides thereof

are on the pinion-gear side and inner-diameter sides thereof
are on the counter-pinion-gear side with respect to the thrust
surface. To be brief, the lines of action are tilted on the
downside in Figs. 2 and 3. The retainers 32 and 33 retain balls
5   30 and 31 respectively constituting the rows of balls 28 and
29 at a position with circumferentially equal intervals.

[33]   Thus, the inner-diameter sides of the lines of action
$\gamma 1$ and $\gamma 2$ of the first double row ball bearing 10 are on the
pinion-gear side with respect to the thrust surface, while the
10   outer-diameter sides of the lines of action $\gamma 3$ and $\gamma 4$ of the
second double row ball bearing 25 are on the pinion-gear side
with respect to the thrust surface, so that the gradients of
the lines of action in accordance with the contact angles of
the bearings 10 and 25 are thereby reverse to each other. Such
15   a constitution is adopted in order to reverse the directions
where the preload is imparted in the bearings 10 and 25.

[34]   In the second double row ball bearing 25, a diameter
of the ball 30 in the small-diameter-side row of balls 28 and
a diameter of the ball 31 in the large-diameter-side row of
20   balls 29 are equal to each other, while pitch circle diameters
D3 and D4 of the respective rows of balls 28 and 29 are different
to each other. More specifically, the pitch circle diameter
D3 of the large-diameter-side row of balls 28 is set to a value
smaller than that of the pitch circle diameter D4 of the
25   small-diameter-side row of balls 29. As described, the second
double row ball bearing 25 has a double row structure (rows
of balls 28 and 29) in which the two rows of balls have the
different pitch circle diameters D3 and D4 to each other.

[35]   As shown in an enlarged view of Fig. 3, the balls 30
30   of the small-diameter-side row of balls 28 are placed to space
at a predetermined radial clearance $\alpha 2$ between the small
diameter outer ring raceway 12a and the small diameter inner
ring raceway 14a in the initial state before assembling into

the differential device. The balls 31 of the large-diameter-side row of balls 29 are placed to space at a predetermined radial clearance $\beta 2$ smaller than the radial clearance $\alpha 2$ ($\alpha 2 > \beta 2$) between the large diameter outer ring raceway 12b and the large diameter inner ring raceway 14b in the initial state before assembling into the differential device.

[36]   An oil-circulating path 40 is formed between an outer wall of the front case 3 and one side of the annular wall 27A. An oil inlet 41 of the oil circulating path 40 is opened toward a ring-gear-8 side of the oil circulating path 40, while an oil outlet 42 of the oil circulating path 40 is opened toward between the annular walls 27A and 27B.

[37]   The differential device 1 comprises a companion flange 43. The companion flange 43 comprises a barrel part 44 and a flange part 45 formed as united with the barrel part 44.

[38]   The barrel part 44 is externally mounted on the shaft part 9 of the pinion shaft 7 on the other side thereof, namely, on a drive-shaft side (not shown) thereof. The shield 37 is interposed between an end surface of the barrel part 44 and an end surface of the second inner ring 14 of the second double row ball bearing 25.

[39]   An oil seal 46 is arranged between an outer peripheral surface of the barrel part 44 and an inner peripheral surface of an opening of the front case 3 on the other side thereof. A seal protective cap 47 is attached to the other-side opening of the front case 3. The oil seal 46 is covered with the seal protective cap 47. A screw part 48 is formed at an end part of the shaft part 9 on the other side thereof. The screw part 48 is protruded into a central recess part 43a of the flange part 45. A nut 49 is screwed into the screw part 48 so that the first inner ring 13 of the first double row ball bearing 10 and the second inner ring 14 of the second double row ball

bearing 25 are sandwiched between the end surface of the pinion gear 6 and an end surface of the companion flange 43 in the axial-center direction. A predetermined preload is imparted to the first double row ball bearing 10 and the second double row ball bearing 25 via the shield 37 and the plastic spacer 23.

[40]   In the differential device 1 thus constituted, a lubricating oil 50 is reserved in the differential case 2 at a predetermined level L in a state where the operation is halted. The lubricating oil 50 is raised upward by the rotation of the ring gear 8 when the operation starts, travels through the oil circulating path 40 in the front case 3, and is introduced and supplied to upper parts of the first double row ball bearing 10 and the second double row ball bearing 25. Thereby, the lubricating oil 50 circulates in the differential case 2 so as to lubricate the first double row ball bearing 10 and the second double row ball bearing 25.

[41]   Next, a method of assembling the differential device 1 is described referring to a partial sectional view of Fig. 4. In order to assemble the differential device 1, the first double row ball bearing 10 and the second double row ball bearing 25 are assembled in advance. Before the first double row ball bearing 10 is assembled, the radial clearance $\beta 1$ is adjusted to be smaller than the radial clearance $\alpha 1$ as described earlier. More specifically, the respective parts of the first double row ball bearing 10 are formed so as to obtain the foregoing relationship between the clearances, and further, shapes of the respective parts are adjusted so that the clearances in the predetermined state can be obtained in the assembly.

[42]   Before the second double row ball bearing 25 is assembled, an clearance between the small-diameter-side row of balls 28, and the small diameter outer ring raceway 12a and the small diameter inner ring raceway 14a is adjusted so that the radial

clearance $\beta 2$ is smaller than the radial clearance $\alpha 2$ as described earlier. More specifically, the respective parts of the second double row ball bearing 25 are formed so as to obtain the foregoing relationship between the clearances, and further, shapes of the respective parts are adjusted so that the clearance in the predetermined state can be obtained in the assembly.

[43]   After the foregoing adjustments and preparations are made, the first double row ball bearing 10 is disassembled into the first outer ring 11 and the first assembly component 21, and the second double row ball bearing 25 is disassembled into the second outer ring 12 and the second assembly component 22. Then, the first double row ball bearing 10 and the second double row ball bearing 25 are incorporated into the differential device 1. More specifically, the first outer ring 11 and the second outer ring 12 are respectively pressed into the annular walls 27A and 27B. More specifically, in a state where the front case 3 and the rear case 4 are still separated, the first outer ring 11 is incorporated into the front case 3 and further pressed in the axial-center direction from the one-side opening of the front case 3 until it abuts a step part formed on the annular wall 27A. Then, the second outer ring 12 is pressed in the axial-center direction from the other-side opening of the front case 3 until it abuts a step part formed on the annular wall 28B.

[44]   The first assembly component 21 (specifically, first inner ring 13) is inserted through the pinion shaft 7. Then, the first assembly component 21 is incorporated into the pinion shaft 7 so as to locate on the pinion-gear-6 side of the shaft part 9 of the pinion shaft 7.

[45]   The pinion shaft 7 into which the first assembly component 21 is incorporated is inserted through the one-side opening of the front case 3 from the small-diameter side thereof. At

the time, the pinion shaft 7 is inserted so that the balls 18 of the small-diameter-side row of balls 16 of the first assembly component 21 are fitted into the small-diameter outer ring raceway 11b of the first outer ring 11. Further, the pinion

5   shaft 7 is inserted so that the balls 17 of the large-diameter-side row of balls 15 are fitted into the large-diameter outer ring raceway 11a of the first outer ring 11. In order to realize the assembly process described above, the small-diameter-side row of balls 18 is arranged to be closer

10   to a rear side in the direction where the pinion shaft 7 is inserted (the counter-pinion-gear side) than the large-diameter-side row of balls 16.

[46]   Next, the plastic spacer 23 is inserted by externally fitting to the shaft part 9 of the pinion shaft 7 from the

15   other-side opening of the front case 3. Subsequently, the second assembly component 22 (specifically, second inner ring 14) is inserted by externally fitting to the shaft part 9 of the pinion shaft 7 from the other-side opening of the front case 3. In order to realize the foregoing insertion by

20   externally fitting, the small-diameter-side row of balls 28 is arranged to be closer to a rear side in the direction where the pinion shaft 7 is inserted (pinion-gear side) than the large-diameter-side row of balls 29.

[47]   Thereafter, the shield 37 is inserted through the shaft

25   part 9 of the pinion shaft 7 from the other-side opening of the front case 3. Further, the oil seal 46 is fixed on the shaft part 9 of the pinion shaft 7 from the other-side opening of the front case 3. The seal protective cap 47 is mounted on the other-side opening of the front case 3. The barrel part

30   44 of the companion flange 43 is inserted through the seal protective cap 47 so that the end surface of the barrel part 44 abuts the shield 37. Then, the nut 49 is screwed into the screw part 48. Thereby, a thrust load is imparted to the first

double row ball bearing 10 and the second double row ball bearing 25, and a predetermined preload is imparted thereto. The direction to impart the preload id done as below. The preload is imparted to the first double row ball bearing 10 along the direction from the pinion-gear side toward the counter-pinion-gear side, while the preload is imparted to the second double row ball bearing 25 along the direction from the counter-pinion-gear side toward the pinion-gear side. Thus, the preload is imparted to the first and second double row ball bearings 10 and 25 in the reverse directions.

[48]   In the differential device 1, the radial clearance β1 is set to a value smaller than that of the radial clearance α1. Therefore, when the thrust load for imparting the preload is applied to the first double row ball bearing 10, the balls 18 of the small-diameter-side row of balls 16 are fitted into the raceways 11b and 13b at the predetermined contact angle before the balls 17 of the large-diameter-side row of balls 15 are fitted into the raceways 11b and 13b, and thereby the rotation torque is generated.

[49]   In the same way, in the differential device 1, the radial clearance β2 is set to a value smaller than that of the radial clearance α2 in a similar manner. Therefore, when the thrust load for imparting the preload is applied to the second double row ball bearing 25, the balls 31 of the large-diameter-side row of balls 29 are fitted into the raceways 12b and 14b at the predetermined contact angle before the balls 30 of the small-diameter-side row of balls 28 are fitted into the raceways 12a and 14a, and thereby the rotation torque is generated.

[50]   In the first double row ball bearing 10 and the second double row ball bearing 25, the initial rotation torques are obtained as described, and the larger thrust load which is further applied so that the preload at a necessary level is applied. Explanation is given below.

[51]   In the state where the initial rotation torques are generated, as described, the radial clearance $\beta 1$, that is on the side of the smaller clearance, is reduced, and the balls 18 of the small-diameter-side row of balls 16, the small diameter outer ring raceway 11b and the small diameter inner ring raceway 13b are thereby already fitted with respect to one another at the predetermined contact angle in the first double row ball bearing 10, while the radial clearance $\beta 2$, that is on the side of the smaller clearance, is reduced so that the balls 31 of the large-diameter-side row of balls 29 and the raceways 12b and 14b are fitted with respect to one another at the predetermined contact angle in the second double row ball bearing 25.

[52]   When the thrust load is further imparted to the first double row ball bearing 10 and the second double row ball bearing 25 in the described state, the radial clearance $\alpha 1$, that is on the side of the larger clearance, is reduced so that the balls 17 of the large-diameter-side row of balls 15 and the raceways 11a and 13a are fitted with respect to one another at the predetermined contact angle in the first double row ball bearing 10, and thereby the rotation torque is generated. In a similar manner, the radial clearance $\alpha 2$, that is on the side of the larger clearance, is reduced so that the balls 30 of the small-diameter-side row of balls 28 and the raceways 12a and 14a are fitted with respect to one another at the predetermined contact angle in the second double row ball bearing 25, and thereby the rotation torque is generated.

[53]   By slightly shifting the fitting timing in the respective rows of balls as described, the rotation torque is selectively obtained in the row of balls 16 alone in the first double row ball bearing 10, and thereafter the rotation torque resulting from the synthesized rotation torques of the rows of balls 15 and 16 is obtained with a time lag. In a similar

manner, in the second double row ball bearing 25, the rotation torque is selectively obtained in the row of balls 29 alone, and thereafter the rotation torque resulting from the synthesized rotation torques of the rows of balls 28 and 29

5   is obtained with a time lag. Accordingly, a maximum rotation torque thereby obtained is increased, which expands the range of the adjustable torque. The adjustment range of the preload to be set is thereby increased, and the preload control is consequently facilitated.

10   [54]   A graph of Fig. 4 shows a relationship between the thrust load S (preload) imparted to the oblique contact double row ball bearing and the rotation torque T corresponding to the thrust load S. The thrust load applied to the oblique contact double row ball bearing can be known through the measurement

15   of the rotation torque T.

[55]   In the drawing, a broken line 60 shows a result of the conventional oblique contact double row ball bearing (double row ball bearing in which the pitch circle diameters of the respective rows are different to each other), while a solid

20   line 61 shows a result of the first and second oblique contact double row ball bearings 10 and 25 according to the present invention (oblique contact double row ball bearings in which the pitch circle diameters of the respective rows are different to each other). Comparing a tilt of the broken line 60 to that

25   of the solid line 61 to each other, the tilt of the solid line 61 is larger than that of the broken line 60. A reason is described below.

[56]   As described, in the first double row ball bearing 10, the balls 18 of the small-diameter-side row of balls 16 and

30   the raceways 11b and 13b are first fitted with respect to one another so that the initial rotation torque is generated, and thereafter, the balls 17 of the large-diameter-side row of balls 15 and the raceways 11a and 13a are fitted with respect to one

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another so that the rotation torque is further generated.

[57]   In a similar manner, in the second double row ball bearing 25, the balls 31 of the large-diameter-side row of balls 29 and the raceways 12b and 14b are first fitted with respect to one another so that the initial rotation torque is generated, and thereafter, the balls 30 of the small-diameter-side row of balls 28 and the raceways 12a and 14a are fitted with respect to one another so that the rotation torque is further generated.

[58]   As a result, the rotation torque can be largely set and the range of the torque to be set can be increased in comparison to the conventional oblique contact double row ball bearing in which the balls of the both rows are simultaneously fitted into the raceways, and the gradient of the full line 61 is larger than that of the broken line 60.

[59]   Description is given here, for example, to a case where it is tried to obtain the "S2" value as the thrust load S referring to a graph shown in Fig. 5. Because the tilt of the solid line 61 is larger than that of the broken line 60, the adjustment range of the rotation torque T corresponding to the "S2" value is T1 in the broken line 60 (conventional example), while the adjustment rage of the rotation torque T is T2 in the first and second double row ball bearings 10 and 25 according to the present invention, that is, it is T2 > T1. Therefore, when the thrust load S2 is applied so that the same preload is obtained, the range for the adjustment is larger in the first and second double row ball bearings 10 and 25 according to the present invention than in the conventional oblique contact double row ball bearing. As a result, the preload can be accurately and easily applied.

[60]   Consideration is done about a case where the thrust load "S2" to be imparted is in the range from "S1" through "S3" in terms of its tolerance. In this case, the adjustment range of the rotation torque T in the conventional oblique contact

double row ball bearing is T3, while the adjustment range of the rotation torque T is T4 in the first and second double row ball bearings 10 and 25 according to the present invention, that is, it is T4 > T3 as shown in Fig. 5. Thus, even in this case, the first and second double row ball bearings 10 and 25 according to the present invention can achieve the adjustment range wider than that of the conventional double row ball bearing when it is tried to obtain the same preload. As a result, the thrust load S (preload) can be accurately and easily applied.